

# Multi-objective optimization of a combined cooling, heating and power system driven by solar energy



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## ARTICLE INFO

### Article history:

Received 15 May 2014

Accepted 5 October 2014

### Keywords:

CCHP

Solar energy

Organic Rankine cycle

## ABSTRACT

This paper presented a multi-objective optimization of a combined cooling, heating and power system (CCHP) driven by solar energy. The flat-plate solar collector was employed to collect the solar radiation and to transform it into thermal energy. The thermal storage unit was installed to store the thermal energy collected by the collectors to ensure a continuous energy supplement when solar energy was weak or insufficient. The CCHP system combined an organic Rankine cycle with an ejector refrigeration cycle to yield electricity and cold capacity to users. In order to conduct the optimization, the mathematical model of the solar-powered CCHP system was established. Owing to the limitation of the single-objective optimization, the multi-objective optimization of the system was carried out. Four key parameters, namely turbine inlet temperature, turbine inlet pressure, condensation temperature and pinch temperature difference in vapor generator, were selected as the decision variables to examine the performance of the overall system. Two objective functions, namely the average useful output and the total heat transfer area, were selected to maximize the average useful output and to minimize the total heat transfer area under the given conditions. NSGA-II (Non-dominated Sort Genetic Algorithm-II) was employed to achieve the final solutions in the multi-objective optimization of the system operating in three modes, namely power mode, combined heat and power (CHP) mode, and combined cooling and power (CCP) mode. For the power mode, the optimum average useful output and total heat transfer area were 6.40 kW and 46.16 m<sup>2</sup>. For the CCP mode, the optimum average useful output and total heat transfer area were 5.84 kW and 58.74 m<sup>2</sup>. For the CHP mode, the optimum average useful output and total heat transfer area were 8.89 kW and 38.78 m<sup>2</sup>. Results also indicated that the multi-objective optimization provided a more comprehensive solution set so that the optimum performance could be achieved according to different requirements for system.

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## 1. Introduction

Nowadays, energy consumption and carbon emission are becoming increasingly urgent issues. The multi-demand of energy for users has led to plenty efforts made by scientists to integrate multi-energy systems to achieve a higher efficiency. In this situation, the combined cooling, heating and power system (CCHP) has proved to be an efficient way to achieve energy saving and economic saving as well as to reduce the emission of pollutants [1,2]. Absorption chiller, compression refrigerator or ejector chiller can be used in the CCHP system to produce cooling capacity.

A large amount of works focused on CCHP systems have been carried out on the performance optimization considering the cost criterion. The linear programming model and the non-line

programming model were employed to conduct the economic optimization of the CCHP system to obtain the minimum value of the cost of energy [3–5]. Ren et al. [6] developed a mixed integer nonlinear programming model to optimize the annual cost of a residential CCHP system. Tichi et al. [7] employed a particle swarm algorithm to minimize the cost function for different CHP and CCHP systems in an industrial dairy unit. It could be confirmed that the CCHP system was more efficient in energy saving.

Except for the economic index, the CCHP system performance can be additionally determined by thermodynamic and environmental criteria. Several articles also carried out additional analysis from the perspective of thermodynamics or environmental aspects. Chen et al. [8] conducted an exergoeconomic performance optimization of a CCHP system with a closed Brayton cycle. The fossil fuel energy savings and the CO<sub>2</sub> emissions were compared with the optimization results of a conventional system to evaluate the system performance more comprehensively. Oh and Seo et al. [9,10]

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### Nomenclature

|       |                                                                                     |
|-------|-------------------------------------------------------------------------------------|
| $A$   | area, m <sup>2</sup>                                                                |
| $Bo$  | boiling number                                                                      |
| $C_p$ | specific heat at constant pressure, kJ/kg K                                         |
| $D$   | diameter, m                                                                         |
| $E$   | exergy, kW                                                                          |
| $F$   | collector efficiency factor                                                         |
| $F_R$ | heat removal factor                                                                 |
| $h$   | convection heat transfer coefficient, W/m <sup>2</sup> K; enthalpy, kJ/kg           |
| $I$   | hourly radiation, W/m <sup>2</sup>                                                  |
| $m$   | mass flow rate, kg/s                                                                |
| $Nu$  | Nusselt number                                                                      |
| $P$   | pressure, MPa                                                                       |
| $Pr$  | Prandtl number                                                                      |
| $Q$   | heat rate, kW                                                                       |
| $R$   | tilt factor for radiation                                                           |
| $Re$  | Reynolds number                                                                     |
| $S$   | incident solar flux, W/m <sup>2</sup>                                               |
| $T$   | temperature, K; Time, h                                                             |
| $U$   | loss coefficient, W/m <sup>2</sup> K; heat transfer coefficient, W/m <sup>2</sup> K |
| $V$   | volume, m <sup>3</sup>                                                              |
| $W$   | pitch of tube, m; power, kW                                                         |
| $x$   | vapor quality                                                                       |

### Greek letters

|              |                                                                    |
|--------------|--------------------------------------------------------------------|
| $\alpha$     | absorptivity                                                       |
| $\beta$      | Chevron angle of the plates                                        |
| $\gamma$     | ratio of the average useful output to the total heat transfer area |
| $\delta$     | thickness, cm                                                      |
| $\Delta t_m$ | log mean temperature difference between hot side and cold side, K  |
| $\eta$       | efficiency, %                                                      |
| $\lambda$    | thermal conductivity, W/m K                                        |
| $\rho$       | density, kg/m <sup>3</sup>                                         |
| $\tau$       | transmissivity; Time, h                                            |
| $\phi$       | latitude, °                                                        |

### Subscripts

|           |                                              |
|-----------|----------------------------------------------|
| $a$       | ambient; absorption                          |
| $b$       | beam radiation                               |
| $c$       | cold side                                    |
| $cond$    | condenser                                    |
| $cool$    | cooling                                      |
| $CND$     | condenser                                    |
| $d$       | diffuse radiation; daily                     |
| $Eq$      | equivalent                                   |
| $EVP$     | evaporator                                   |
| $h$       | hot side; hydraulic diameter of flow channel |
| $heat$    | heat                                         |
| $heater$  | heater                                       |
| $in$      | input                                        |
| $I$       | inlet; inner                                 |
| $l$       | loss; liquid phase                           |
| $L$       | liquid in the water tank                     |
| $FC$      | flat-plate collector                         |
| $load$    | load                                         |
| $m$       | mean                                         |
| $max$     | maximum                                      |
| $min$     | minimum                                      |
| $net$     | net                                          |
| $out$     | output                                       |
| $o$       | outer                                        |
| $p$       | collector plate                              |
| $pinch$   | pinch point                                  |
| $pump$    | pump                                         |
| $r$       | reflected radiation                          |
| $s$       | isentropic point; Sun                        |
| $solar$   | solar                                        |
| $t$       | tank                                         |
| $tot$     | total                                        |
| $turbine$ | turbine inlet                                |
| $TBN$     | turbine                                      |
| $u$       | useful heat gain                             |
| $v$       | vapor phase                                  |
| $VG$      | vapor generator                              |
| $w$       | water in the tank                            |
| $water1$  | water into the solar collector               |
| $water2$  | water into the vapor generator               |

used the mixed integer linear programming models to minimize the total annual cost of trigeneration systems in commercial and residential sectors. Arcur et al. [11] conducted the economic optimization of a trigeneration system in an energetically complex context by using a mixed linear programming model. Then, the energetics and environmental analysis were conducted based on results of the economic optimization to achieve a better system performance.

In order to evaluate the system performance comprehensively, certain researches focused on the optimization of CCHP systems were performed concerning simultaneously energetic, economic and environmental aspects. The single-objective optimizations for the CCHP system [12–15] were conducted by integrating primary energy consumption (PEC), operation cost (OC), and carbon dioxide emissions (CDE) or other index as the objective function respectively. The results demonstrated that there was not an optimal condition where each objective all reached the optimum value. Liu et al. [16] utilized the sequential quadratic programming (SQP) algorithm to solve the optimization problem of a CCHP system. The ratio of electric cooling to cool load was varied to minimize the

objective function which was made up of three parts, namely the primary energy consumption, the hourly and annually total cost and the greenhouse gas (GHG) emission.

However, much work has been done on the single-objective optimization of the CCHP system, while few studies have been investigated in the multi-objective optimization. Abdollahi and Meratizaman [17–19] used the genetic algorithm to conduct the multi-objective optimization of a CCHP system. The objectives functions were selected from the view of thermodynamic, economic and environmental aspects. A two-stage optimal planning and design method was applied for a CCHP system [20]. Non-dominated Sorting Genetic Algorithm-II (NSGA-II) was employed in multi-objective optimization on the first stage, and the mixed-integer linear programming algorithm on the second stage.

All of these researches above have been devoted to CCHP powered by fossil fuels which will be limited with the growing crisis of energy shortage and are harm to environment with CO<sub>x</sub> and NO<sub>x</sub> emissions. The renewable energies, including hydraulic energy, hydrogen energy, solar energy and so on, were clean and inexhaustible, which would contribute to resolve the problem of

energy crisis and emission of pollutions. In this situation, integrating the renewable energy technology to the CCHP system seems to be an efficient way to improve the energy efficiency and to reduce the pollutant emission to the environment. Solar energy, as a kind of renewable energy resources, is widely applied to the trigeneration system [21]. Compared with the independent cooling, heating and power systems powered by solar energy, the solar-powered CCHP system has attracted more attention of researchers. Because in the combined configuration, the three subsystems can share same solar collection and storage system, pipes or auxiliary equipment, while the independent systems require respective independent equipment which may cause a higher cost. Wang et al. [22] combined a Brayton cycle and a transcritical CO<sub>2</sub> refrigeration cycle to perform the CCHP system driven by solar energy. Furthermore, they [23] also examined a solar-powered CCHP system for a specific building based on organic working fluid. Parametric analysis was conducted to track the operation performance and to analyze the influence of key parameters on the performance from both thermodynamic and economic aspects.

From the view of engineering application, it is necessary to conduct the optimization of solar-powered CCHP which is helpful to the design of system. Wang et al. [24,25] proposed a new CCHP system powered by solar energy. Genetic algorithm was employed to carry out the system optimization with modified system efficiency or the exergy efficiency as the objective function respectively. Tora and El-Halwagi [26] used a nonlinear programming formulation to minimize the total annualized cost of a trigeneration system driven by solar energy and fossil fuels. Since there are many factors having the significant effects on the system performance from thermodynamic, economic and environmental aspects, the single-objective optimization could not evaluate the performance of the solar-powered CCHP system comprehensively. Multi-objective optimization should be urgent to be conducted.

In this study, the multi-objective optimization of a combined cooling, heating and power system (CCHP) driven by solar energy is conducted to obtain the optimum performance from the view of two aspects. Non-dominated Sorting Genetic Algorithm-II is employed to conduct the multi-objective optimization. The trigeneration system is self-sufficient by using flat-plate collectors, meeting heating, cooling and power demands of users simultaneously. A thermal storage tank is employed to store solar energy collected by collectors and to supply energy to the CCHP subsystem continuously even when the solar radiation is insufficient. The average useful output and the total heat transfer area are selected as the

objective functions. Turbine inlet temperature, turbine inlet pressure, condensation temperature and pinch temperature difference in vapor generator are selected as the decision variables.

## 2. System description

The CCHP system powered by solar energy consists of two subsystems, namely the solar collection subsystem and the CCHP subsystem, as shown in Fig. 1.

The solar collection subsystem is made up of a solar collector, a thermal storage tank and an auxiliary heater. The flat-plate collector is selected to collect solar radiation due to its low cost and wide application. A thermal storage tank is employed to correct the mismatch between the supply of the solar energy and the demand of thermal source consumed by the CCHP subsystem, thus the system could operate continuously and stably. On rainy days or at night, when the solar radiation is insufficient and the temperature of thermal storage tank drops below the allowable reference temperature, the auxiliary heater is activated to elevate the outlet temperature of thermal storage tank. The heat-transfer medium in the solar collection subsystem is water for its low cost and large heat capacity.

The CCHP subsystem, which mainly combines an organic Rankine cycle (ORC) with an ejector refrigeration cycle, consists of a vapor generator, a turbine, an evaporator, a heater, a condenser, a recuperator, a throttle valve, an ejector, a pump, and several regulation valves. In the vapor generator, the liquid working fluid with high pressure is vaporized by absorbing heat from the hot water, and then the high pressure vapor is divided into three streams. One stream is delivered to a turbine where it expands to a low pressure to drive a generator to produce electricity. Another stream flows into an ejector as a primary flow to suck the secondary flow from an evaporator with very low pressure. Afterwards, the primary flow and the secondary flow are mixed together to a mid-pressure leaving the ejector. The third stream is delivered to a heater to provide heat energy to users and then flows into a recuperator. The stream leaving the ejector is mixed with the turbine exhaust and then condensed to liquid state in a condenser by rejecting heat to the environment. The liquid working fluid is also divided into two streams. One stream is throttling down to a very low pressure across a throttle valve and then is vaporized in an evaporator by absorbing heat to produce cooling effect. The other stream is pumped to a high pressure and enters a recuperator where it is mixed with the stream out of the heater.

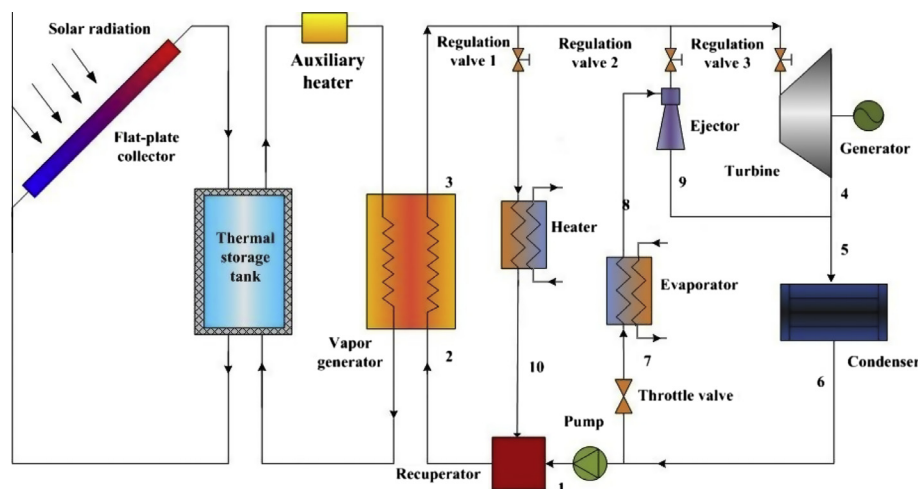


Fig. 1. Schematic diagram of the solar-powered CCHP system.

Finally, the working fluid leaving the recuperator is delivered back to the vapor generator.

In this case, the CCHP system can operate in three modes, namely power mode, combined heat and power (CHP) mode, and combined cooling and power (CCP) mode. During a year-round, power load is needed by the users all the time. In spring or autumn, heat and cooling is not necessary so that the system operates in power mode by closing the regulation valve 1 and 2. In winter, regulation valve 1 is opened to make the system operate in CHP mode. In summer, regulation valve 2 is opened while the regulation valve 1 is closed so that the system could discharge cooling and power to load in CCP mode.

As to the working fluid in the CCHP subsystem, we consider thermodynamic properties and environmental friendliness, and finally select R245fa as the working fluids of the system due to its good performance and little negative impact on environment.

### 3. Mathematical models and performance criteria

#### 3.1. Mathematical models

The mathematical models of the solar-powered CCHP system mainly consist of the solar collection models and the CCHP models.

The flat-plate collector is selected to absorb the solar radiation. The hourly radiation falling on a tilted surface can be given by

$$I_{FC} = I_b R_b + I_d R_d + (I_b + I_d) R_r \quad (1)$$

where  $I_b$  and  $I_d$  are the hourly beam and diffuse radiation that the collector receives,  $R_b$ ,  $R_d$ , and  $R_r$  are defined as tilt factors for different radiations. The solar radiation and the ambient temperature are assumed to be constant within one hour.

The incident solar flux absorbed by the absorber plate is calculated by

$$S = I_b R_b (\tau\alpha)_b + [I_d R_d + (I_b + I_d) R_r] (\tau\alpha)_d \quad (2)$$

where  $(\tau\alpha)_b$  and  $(\tau\alpha)_d$  represent the transmissivity-absorptivity product falling on the collector.

Considering the heat loss in the collector, the useful heat gain rate of the collectors can be achieved by introducing the collector heat-removal factor  $F_R$  [27]

$$F_R = \frac{m_{\text{water}1} C_p}{U_1 A_p} \left[ 1 - \exp \left( - \frac{F' U_1 A_p}{m_{\text{water}1} C_p} \right) \right] \quad (3)$$

$$Q_u = F_R \cdot A_p \cdot [S - U_1 \cdot (T_L - T_a)] \quad (4)$$

where the collector efficiency  $F'$  is determined by [28]

$$F' = \frac{1}{W U_1 \left\{ \frac{1}{U_1 [(W-D)\phi + D_o]} + \frac{\phi_o}{k_a D_o} + \frac{1}{\pi D_1 h_f} \right\}} \quad (5)$$

A water storage tank is installed to act as a buffer between the solar collector and the bottom CCHP subsystem. The tank is assumed to be insulated and the water is assumed to be well-mixed so that the water temperature  $T_L$  varies only with time. The following equation can be obtained from the energy balance in the tank [28]

$$[(\rho V C_p)_w + (\rho V C_p)_L] \frac{dT_L}{dt} = Q_u - Q_{\text{load}} - (U A)_t \cdot (T_L - T_a) \quad (6)$$

Furthermore,  $Q_{\text{load}}$ , which represents the energy discharged to CCHP subsystem, can be calculated as

$$Q_{\text{load}} = m_{\text{water}2} C_p (T_L - T_i) \quad (7)$$

The bottom CCHP subsystem combines an organic Rankine cycle with an ejector refrigeration cycle, yielding heat, power and cooling to users. To simplify the simulation of the system, several assumptions about the CCHP subsystem are made [24]:

- (1) The pressure drop in pipes or other components, the heat loss to the environment in the heat exchangers, and the frictional and mixing losses in the ejector are neglected.
- (2) The working fluid at the condenser outlet is saturated liquid. The condensation temperature is approximately 5 °C higher than the environment temperature.
- (3) The flow through the throttle valve is isenthalpic and the flow in ejector is assumed to be one-dimensional.

For the vapor generator, the heat balance equation is:

$$C_p m_{\text{water}2} (T_L - T_i) = m_3 (h_3 - h_2) \quad (8)$$

For the turbine and the pump, the isentropic efficiency is considered. The power generated by the turbine and consumed by the pump are the assumed to be the power transferred from the fluid to the moving parts of turbine and to the fluid from the moving parts of pump respectively.

$$\eta_{\text{TBN}} = \frac{h_3 - h_4}{h_3 - h_{4s}} \quad (9)$$

$$W_{\text{TBN}} = m_4 (h_3 - h_4) \quad (10)$$

$$\eta_{\text{pump}} = \frac{h_{1s} - h_6}{h_1 - h_6} \quad (11)$$

$$W_{\text{pump}} = m_1 (h_1 - h_6) \quad (12)$$

Therefore, the net power output can be given by

$$W_{\text{net}} = W_{\text{TBN}} - W_{\text{pump}} \quad (13)$$

For the heater and evaporator, the heat and cooling output to users respectively are:

$$Q_{\text{heat}} = m_{10} \cdot (h_3 - h_{10}) \quad (14)$$

$$Q_{\text{cool}} = m_8 \cdot (h_8 - h_7) \quad (15)$$

All the heat exchangers in this system are plate heat exchanger (PHE) type for its high efficiency and compact structure. The total heat transfer area of the CCHP system is determined from a summation of individual components:

$$A_{\text{tot}} = A_{\text{CND}} + A_{\text{VG}} + A_{\text{EVP}} + A_{\text{Heater}} \quad (16)$$

In power mode, the heater and the evaporator do not work, so the heat transfer area of the heater and the evaporator are not considered in the calculation of the total heat transfer area. While in CCP mode the heat transfer area of the heater is not considered, and in CHP mode the heat transfer area of the evaporator is not considered.

In the heat exchanger, the working fluid may work in different thermodynamic states, namely superheated state, two-phase state and subcooled state, resulting in that the heat transfer process in the heat exchanger can be divided into different regions. In the condenser, the working fluid is liquefied from superheated state to saturated liquid state undergoing the superheated region and the two-phase region. In the vapor generator and the heater, the working fluid respectively works in subcooled region, two-phase region and superheated region. In the vapor generator, the pinch temperature difference is an important parameter, which is the minimum temperature difference between the heat source and the saturation temperature of working fluid at the evaporation pressure. In the evaporator, however, the working fluid is vaporized only in two-phase region. Compared to the two-phase region, the superheated region and the subcooled region are single-phase region.

In the heat exchanger, the general formula of the heat rate in any region can be given by [29]

$$Q = U A \Delta t_m \quad (17)$$

The log mean temperature difference is expressed as

$$\Delta t_m = \frac{\Delta t_{\max} - \Delta t_{\min}}{\ln(\Delta t_{\max}/\Delta t_{\min})} \quad (18)$$

The overall heat transfer coefficient is given by

$$\frac{1}{U} = \frac{1}{h_h} + \frac{\delta}{\lambda_m} + \frac{1}{h_c} \quad (19)$$

where the convection heat transfer coefficient for single-phase flow is expressed as

$$h = \frac{\lambda \cdot \text{Nu}}{D_h} \quad (20)$$

For the single phase region, the Nusselt number is calculated by using the Chisholm and Wanniarachchi correlation

$$\text{Nu} = 0.724 \left( \frac{6\beta}{\pi} \right)^{0.646} \text{Re}^{0.583} \text{Pr}^{1/3} \quad (21)$$

where  $\beta$  stands for the chevron angle of the plates, being expressed by Ref. [30].

For the two-phase region, the heat transfer process is divided into relatively small sections so that the properties in each section can be assumed to be constant with such slight property variations. The Nusselt numbers are different when the working fluid is vaporized or condensed. For each section, the calculations of Nusselt number [31,32] in the hot side and in the cold side when the working fluid is in two-phase region can be respectively given by

$$\text{Nu}_{i,h} = 4.118 \text{Re}_{eq,i}^{0.4} \text{Pr}_l^{1/3} \quad (22)$$

$$\text{Nu}_{i,c} = 1.926 \text{Re}_{eq,i}^{0.5} \text{Pr}_l^{1/3} \text{Bo}_{eq,i}^{0.3} \left[ 1 - x_{m,i} + x_{m,i} \left( \frac{\rho_l}{\rho_v} \right)^{0.5} \right] \quad (23)$$

In addition, the one-dimensional model of the ejector is used to simplify the simulation process. The detailed description about the ejector model is given in Ref. [33].

### 3.2. Performance criteria

Since the quality of cold energy, heat energy and power energy are different, the sum of cold, heat and power could not reflect the output capacity of the whole system. Thus, two factors, namely  $\alpha$  and  $\beta$ , are introduced to modify the different price between cold, heat and power energy of the output of the system, given by

$$W_{\text{out}} = W_{\text{net}} + \alpha Q_{\text{heat}} + \beta Q_{\text{cold}} \quad (24)$$

where  $\alpha$  and  $\beta$  vary with the price fluctuation of cooling, heating and power. In this paper,  $\alpha = 0.5$  and  $\beta = 0.8$  [34].

The working condition of the system varies with time over a day. A daily average efficiency of the CCHP system is modified compared to the instantaneous efficiency to assess the whole-day performance of the system, shown as

$$\eta_d = \frac{\int W_{\text{out}} d\tau}{\int Q_u d\tau} \quad (25)$$

The daily exergy efficiency is examined from the view of the second law of thermodynamics, shown as

$$\eta_{\text{ex}} = \frac{\int W_{\text{out}} d\tau}{\int E_{\text{in}} d\tau} \quad (26)$$

where  $E_{\text{in}}$  represents the input exergy of the system.

In this CCHP system,  $E_{\text{in}}$  can be calculated as

$$E_{\text{in}} = E_{\text{solar}} \quad (27)$$

where  $E_{\text{solar}}$  is the exergy input of the solar collector, defined as

$$E_{\text{solar}} = A_p I_{\text{FC}} \left[ 1 + \frac{1}{3} \left( \frac{T_a}{T_s} \right)^4 - \frac{4}{3} \left( \frac{T_a}{T_s} \right) \right] \quad (28)$$

$T_s$  is the sun temperature and equals to 6000 K according to Ref. [35].

In order to assess the system performance from both thermodynamic and economic aspects, a thermo-economic index  $\gamma$ , named as output-to-area ratio, is defined as the ratio of the average useful output to the total heat transfer area of the CCHP system in the single-objective optimization, calculated by

$$\gamma = \frac{W_{\text{out,m}}}{A_{\text{tot}}} \quad (29)$$

where  $W_{\text{out,m}}$  presents the average useful output of the CCHP system over a day

$$W_{\text{out,m}} = \frac{1}{T} \int W_{\text{out}} d\tau \quad (30)$$

## 4. Results and discussions

In order to achieve the optimal performance of the solar-powered CCHP system, we conduct the optimization in the three typical conditions year-round. The Genetic Algorithm (GA) is employed to find the optimum performance of system. Four key thermodynamic parameters, namely turbine inlet temperature, turbine inlet pressure and condensation temperature and pinch temperature difference are chosen as the decision variables. The main thermodynamic parameters for the simulation of the solar-powered CCHP system are listed in Table 1.

The ranges of the decision variables in the optimization are listed in Table 2. The thermodynamic properties of the working fluids are calculated by REFPROP 9.01 [36] developed by the National Institute of Standards and Technology of the United States.

### 4.1. Single-objective optimization

For the single-objective optimization, the objective function is the defined thermo-economic index,  $\gamma$ . The optimum system performance and the corresponding combination of the decision variables are shown in Table 3. The results demonstrate that the optimum turbine inlet temperature in power, CCP and CHP mode are 77.59 °C, 78.01 °C and 67.63 °C respectively, and the turbine inlet pressure is approximately equal to the corresponding saturated pressure. Because the cooling and power are both high quality energies, generating power or cooling energy needs to consume more energy. Therefore, the optimal turbine inlet temperatures of

**Table 1**  
Simulation conditions for the solar-powered CCHP system.

| Term                                              | Value         |
|---------------------------------------------------|---------------|
| Simulation date in winter                         | December 21st |
| Simulation date in summer                         | July 20th     |
| Simulation date in spring/autumn                  | March 21st    |
| Collector tilt angle (CHP mode)/°                 | 61            |
| Collector tilt angle (CCP mode)/°                 | 7             |
| Collector tilt angle (power mode)/°               | 35            |
| Outer diameter of the absorber tube/m             | 0.018         |
| Inner diameter of the absorber tube/m             | 0.014         |
| Surface area of one absorber plate/m <sup>2</sup> | 1.76          |
| Number of absorber plates                         | 500           |
| Number of water tank                              | 20            |
| Evaporator temperature/°C                         | 5             |
| Heater outlet temperature/°C                      | 50            |
| Turbine isentropic efficiency/%                   | 60            |
| Pump isentropic efficiency/%                      | 70            |



**Table 2**

Data of the parameter optimization.

| Term                                     | Value            |
|------------------------------------------|------------------|
| Population size                          | 60               |
| Crossover probability                    | 0.8              |
| Mutation probability                     | 0.02             |
| Stop generation                          | 100              |
| Range of turbine inlet temperature/°C    | 65–80            |
| Range of turbine inlet pressure/MPa      | 0.40–0.80        |
| Range of condensation temperature/°C     | CCP mode 35–45   |
|                                          | Power mode 15–25 |
|                                          | CHP mode 10–20   |
| Range of pinch temperature difference/°C | 5–10             |

**Table 3**

Single-objective optimization results of the solar-powered CCHP system in three operation modes.

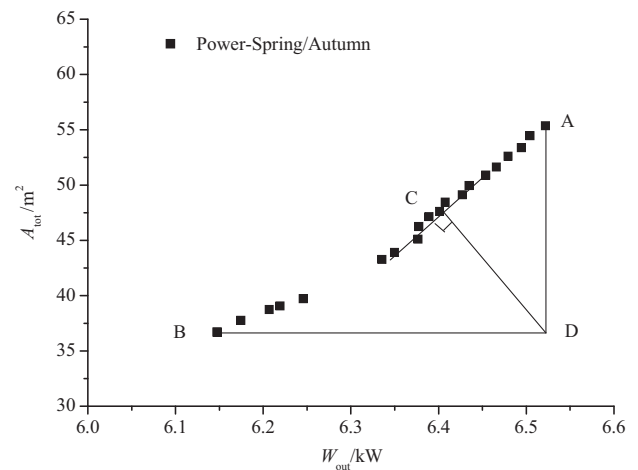
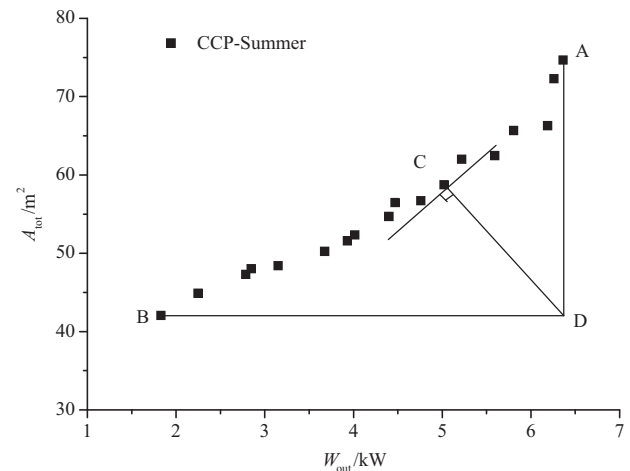
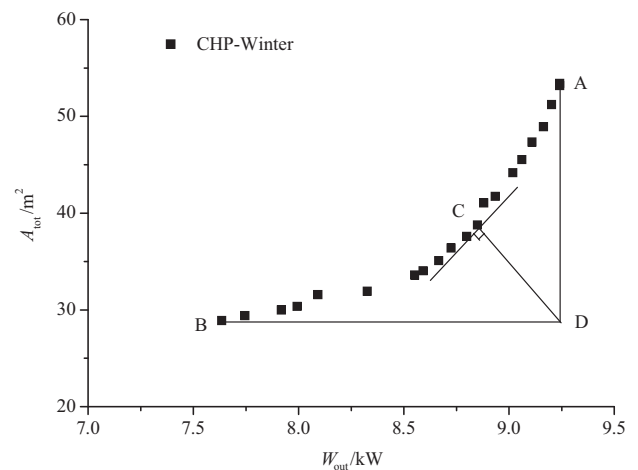
| Term                                    | Power mode | CCP mode | CHP mode |
|-----------------------------------------|------------|----------|----------|
| Output-to-area ratio/kW m <sup>-2</sup> | 0.191      | 0.125    | 0.270    |
| Average useful output/kW                | 6.96       | 7.16     | 8.45     |
| Total heat transfer area/m <sup>2</sup> | 36.43      | 62.76    | 31.34    |
| Daily average efficiency/%              | 9.75       | 8.76     | 13.81    |
| Daily exergy efficiency/%               | 6.16       | 6.16     | 8.41     |
| Turbine inlet temperature/°C            | 77.59      | 78.01    | 67.63    |
| Turbine inlet pressure/MPa              | 0.743      | 0.751    | 0.572    |
| Condensation temperature/°C             | 20.00      | 35.00    | 10.00    |
| Pinch temperature difference/°C         | 10.00      | 10.00    | 9.97     |

system in power mode and in CCP mode are higher than that of system in CHP mode. In addition, the optimum condensation temperatures are close to their minimum values while the optimum values of pinch temperature difference is close to their maximum values in their design ranges. The optimal system performances under this condition of optimization are 0.191 kW/m<sup>2</sup> for power mode, 0.125 kW/m<sup>2</sup> for CCP mode and 0.270 kW/m<sup>2</sup> for CHP mode, meanwhile the exergy efficiencies in power, CCP and CHP mode reach 6.16%, 6.16% and 8.41% respectively.

#### 4.2. Multi-objective optimization

Non-dominated Sorting Genetic Algorithm-II (NSGA-II), which is an effective multi-objective optimization method, is employed to find the solution of the multi-objective optimization. Owing to the fluctuation of solar energy over a day, the useful output of system varies correspondingly. Therefore, the average useful output is defined to assess the capacity of the system. Expect for thermodynamic performance, the economic index should be another important criterion to examine the performance of the CCHP system. Actually, the size of turbine is independent on the power capacity to some degree because larger capacity turbine can use high rotational speed design to decrease its size. Thus, the size of turbine is not sensitive to the capacity which is affected by thermodynamic parameters. So is the pump. But the areas of heat exchangers are very sensitive to thermodynamic parameters. As the heat transfer area is the mainly effect factor of the capital cost of the heat exchanger, the total heat transfer area, which consists of the heat transfer area of vapor generator, evaporator, condenser and heater, is selected as another performance criterion. Eventually, the objectives of the multi-objective optimization are to maximize the average useful output  $W_{out, m}$  and to minimize the total heat transfer area  $A_{tot}$ .

Figs. 2–4 show the Pareto frontier solutions for the solar-powered CCHP system in three operation modes of multi-objective optimization, which reveal the conflicts between average useful output and total heat transfer area. Comparison among the three figures demonstrates that the total heat transfer area in

**Fig. 2.** Results of multi-objective optimization in power mode.**Fig. 3.** Results of multi-objective optimization in CHP mode.**Fig. 4.** Results of multi-objective optimization in CCP mode.

CHP mode is smaller than those in other two modes. The reason is that the cooling and power are both high quality energy which requires a larger total heat transfer area in a whole system for same capacity output. However, the larger average useful output happens to the system in CCP mode. Because the solar radiation

in summer is much stronger than those in other seasons, implying that the system receives more heat input. Results also indicate that in order to achieve a large average useful output, the total heat transfer area should be large enough as well, while the total heat transfer area increases rapidly when the average useful output is greater than the specific value. Therefore, only seeking for a large average useful output is not the best choice for the optimal system design.

In these figures, each point located on the Pareto frontier is potentially an optimum solution. A larger average useful output requires a larger heat transfer area. The maximum average useful output is defined as design point A, where the total heat transfer area is at its maximum value as well. The minimum total heat transfer area is defined as design point B, with corresponding minimum average useful output. Design point A and B respectively imply the optimum solutions of single-objective function where the average useful output or the total heat transfer area is selected as the objective function. Since it is impossible to make each objective at its optimum value simultaneously, the absolutely optimum solution of multi-objective does not exist. The point satisfying the result with the minimum total heat transfer and the maximum average useful output simultaneously does not exist in the optimal solutions, shown as Point D in Figs. 2–4. Therefore, in this paper, the point which is the closest point to the specific point D, is decided to be the final solution in the multi-objective optimization. Thus, to achieve the final solution, a process of decision-making is required with the aid of a hypothetical point D [37] which is the intersection of maximum output and maximum heat transfer area and is not located on the Pareto frontier. In this study, the closest point C of the Pareto frontier to the hypothetical point D is considered as the final solution. At point C, the system achieves the best possible values considering two objective functions. Tables 4–9 list the optimum objective function values and the corresponding decision variables for points A–C on the Pareto frontier and the corresponding detailed parameter distribution on the system in power, CHP and CCP modes respectively. Results indicate that under the condition of the optimum solutions of the multi-objective optimization of the combined system in CCP and CHP mode, the cooling-to-power ratio and heat-to-power ratio are 1.29 and 1.01 respectively.

Fig. 5 depicts the distribution of turbine inlet pressure for the Pareto frontier in its design range in different modes. It can be observed that the optimum solutions of turbine inlet pressure in CCP mode and CHP mode respectively locate in the range of 0.5–0.75 MPa, while in power mode the solutions are around 0.7 MPa.

Fig. 6 depicts the distribution of turbine inlet temperature for the Pareto frontier in its design range in different modes. It is shown that the optimum solutions of turbine inlet temperature locate in the range of 70–78 °C in CCP mode and CHP mode and 75–77 °C in power mode respectively. The result of the optimum turbine inlet temperature is corresponded to that of the optimum turbine inlet pressure. In summer when the solar radiation is

**Table 4**

Optimum design objective values and corresponding decision variables for points A–C on the Pareto optimal front in power mode.

| Term                                    | A     | B     | C     |
|-----------------------------------------|-------|-------|-------|
| Average useful output/kW                | 6.52  | 6.15  | 6.40  |
| Total heat transfer area/m <sup>2</sup> | 50.60 | 38.45 | 46.16 |
| Daily average efficiency/%              | 9.61  | 9.61  | 9.54  |
| Daily exergy efficiency/%               | 6.52  | 6.15  | 6.40  |
| Turbine inlet temperature/°C            | 75.04 | 76.44 | 75.79 |
| Turbine inlet pressure/MPa              | 0.695 | 0.721 | 0.689 |
| Condensation temperature/°C             | 20.00 | 20.03 | 20.04 |
| Pinch temperature difference/°C         | 5.43  | 9.95  | 6.76  |

**Table 5**

Detailed parameter distribution on the system in power mode.

| State | t/°C  | p/MPa | $\rho/\text{kg m}^{-3}$ | h/kJ kg <sup>-1</sup> | s/kJ kg <sup>-1</sup> K <sup>-1</sup> | m/kg s <sup>-1</sup> |
|-------|-------|-------|-------------------------|-----------------------|---------------------------------------|----------------------|
| 1     | 20.37 | 0.721 | 1352.81                 | 226.53                | 1.092                                 | 0.426                |
| 2     | 20.37 | 0.721 | 1352.81                 | 226.53                | 1.092                                 | 0.426                |
| 3     | 76.41 | 0.721 | 40.08                   | 459.39                | 1.778                                 | 0.426                |
| 4     | 35.55 | 0.123 | 6.70                    | 433.65                | 1.799                                 | 0.426                |
| 5     | 35.55 | 0.123 | 6.70                    | 433.65                | 1.799                                 | 0.426                |
| 6     | 20.03 | 0.123 | 1351.94                 | 225.90                | 1.091                                 | 0.426                |

**Table 6**

Optimum design objective values and corresponding decision variables for points A–C on the Pareto optimal front in CCP mode.

| Term                                    | A     | B     | C     |
|-----------------------------------------|-------|-------|-------|
| Average useful output/kW                | 7.40  | 2.13  | 5.84  |
| Total heat transfer area/m <sup>2</sup> | 74.66 | 42.07 | 58.74 |
| Daily average efficiency/%              | 8.71  | 2.26  | 6.90  |
| Daily exergy efficiency/%               | 6.36  | 1.83  | 5.02  |
| Turbine inlet temperature/°C            | 77.68 | 70.65 | 77.17 |
| Turbine inlet pressure/MPa              | 0.744 | 0.499 | 0.694 |
| Condensation temperature/°C             | 35.11 | 42.36 | 37.72 |
| Pinch temperature difference/°C         | 6.90  | 9.76  | 9.61  |

**Table 7**

Detailed parameter distribution on the system in CCP mode.

| State | t/°C  | p/MPa | $\rho/\text{kg m}^{-3}$ | h/kJ kg <sup>-1</sup> | s/kJ kg <sup>-1</sup> K <sup>-1</sup> | m/kg s <sup>-1</sup> |
|-------|-------|-------|-------------------------|-----------------------|---------------------------------------|----------------------|
| 1     | 38.01 | 0.694 | 1304.12                 | 249.98                | 1.170                                 | 0.527                |
| 2     | 38.01 | 0.694 | 1304.12                 | 249.98                | 1.170                                 | 0.527                |
| 3     | 77.17 | 0.694 | 38.06                   | 460.97                | 1.784                                 | 0.527                |
| 4     | 50.52 | 0.232 | 12.41                   | 444.82                | 1.797                                 | 0.293                |
| 5     | 55.36 | 0.232 | 12.16                   | 449.62                | 1.811                                 | 0.557                |
| 6     | 37.72 | 0.232 | 1303.24                 | 249.47                | 1.169                                 | 0.557                |
| 7     | 5.00  | 0.066 | 18.49                   | 249.47                | 1.178                                 | 0.030                |
| 8     | 5.00  | 0.066 | 3.99                    | 408.12                | 1.748                                 | 0.030                |
| 9     | 60.72 | 0.232 | 11.91                   | 454.93                | 1.827                                 | 0.264                |

**Table 8**

Optimum design objective values and corresponding decision variables for points A–C on the Pareto optimal front in CHP mode.

| Decision variables                      | A     | B     | C     |
|-----------------------------------------|-------|-------|-------|
| Average useful output/kW                | 9.28  | 7.67  | 8.89  |
| Total heat transfer area/m <sup>2</sup> | 53.17 | 28.90 | 38.78 |
| Daily average efficiency/%              | 13.63 | 13.91 | 13.59 |
| Daily exergy efficiency/%               | 9.24  | 7.63  | 8.85  |
| Turbine inlet temperature/°C            | 71.91 | 76.69 | 72.55 |
| Turbine inlet pressure/MPa              | 0.802 | 0.716 | 0.513 |
| Condensation temperature/°C             | 10.05 | 10.26 | 10.11 |
| Pinch temperature difference/°C         | 5.14  | 9.99  | 8.03  |

**Table 9**

Detailed parameter distribution on the system in CHP mode.

| State | t/°C  | p/MPa | $\rho/\text{kg m}^{-3}$ | h/kJ kg <sup>-1</sup> | s/kJ kg <sup>-1</sup> K <sup>-1</sup> | m/kg s <sup>-1</sup> |
|-------|-------|-------|-------------------------|-----------------------|---------------------------------------|----------------------|
| 1     | 10.34 | 0.513 | 1303.24                 | 213.42                | 1.047                                 | 0.322                |
| 2     | 15.66 | 0.513 | 1364.68                 | 220.32                | 1.071                                 | 0.371                |
| 3     | 72.55 | 0.513 | 27.16                   | 460.42                | 1.799                                 | 0.371                |
| 4     | 33.67 | 0.082 | 4.46                    | 433.21                | 1.821                                 | 0.322                |
| 5     | 33.67 | 0.082 | 4.46                    | 433.21                | 1.821                                 | 0.322                |
| 6     | 10.11 | 0.082 | 1378.06                 | 212.98                | 1.047                                 | 0.322                |
| 10    | 50.00 | 0.513 | 1268.18                 | 266.32                | 1.222                                 | 0.048                |

strong, the working fluid can be heated to a higher evaporation temperature.

Fig. 7 depicts the distribution of condensation temperature for the Pareto frontier in its design range in different modes. It is dem-

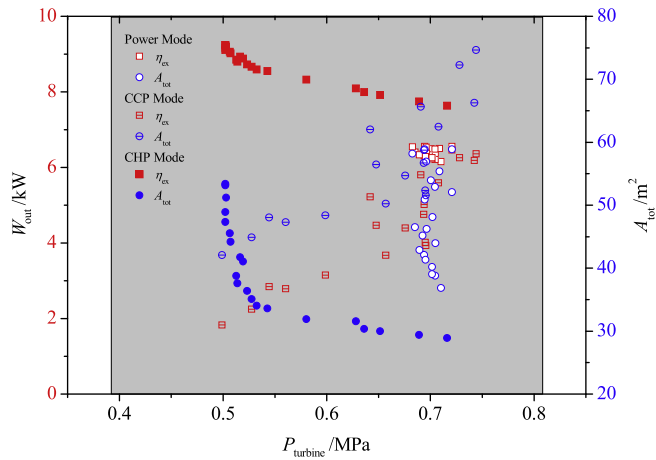


Fig. 5. Distribution of turbine inlet pressure for the Pareto frontier in its design range in different modes.

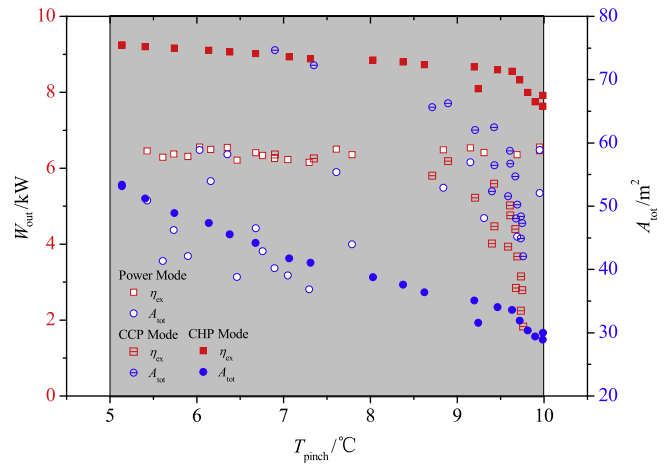


Fig. 8. Distribution of pinch temperature difference for the Pareto frontier in its design range in different modes.

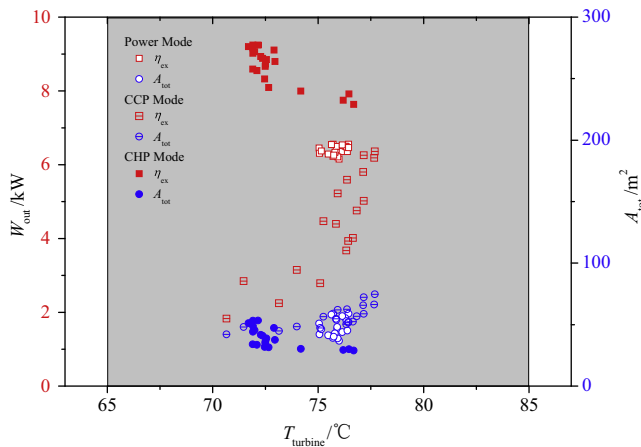


Fig. 6. Distribution of turbine inlet temperature for the Pareto frontier in its design range in different modes.

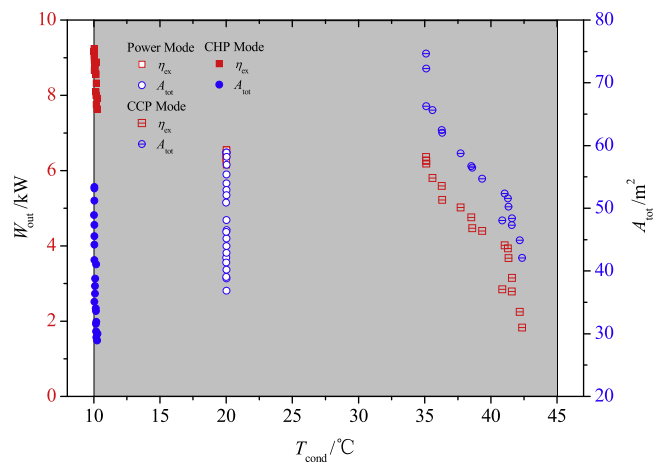


Fig. 7. Distribution of condensation temperature for the Pareto frontier in its design range in different modes.

onstrated that the optimal solutions of condensation temperature in power mode and CHP mainly locates in around the minimum value of the design ranges, and the optimum solutions of turbine inlet pressure in CCP mode exist in the range of 35–43 °C. It can

be concluded that the optimum condensation temperature is not always the smaller ones. The condensation temperature affects the heat transfer of not only condenser but also other heat exchangers in the system. The optimum solution would be various in different conditions.

Fig. 8 depicts the distribution of pinch temperature difference for the Pareto frontier in different modes. The optimum pinch temperature difference in power mode and CHP mode locate in the range across the design range, meanwhile the optimum pinch temperature difference in CCP locates in the range of 7–10 °C.

In general, the heat source temperature has a significant impact on the turbine inlet temperature, while system in CHP mode requires for a lower heat source temperature. Hence, most of the optimal turbine inlet temperature solutions for system in CHP mode are lower than other modes. As to the condensation temperature, it is obvious that the lower condensation temperature, the more power output, while the heat and cooling are barely affected. However the heat transfer area of condenser significantly increases when the condensation temperature is set to a low value. Finally, the optimal solutions of condensation temperature in power mode locate in the specific range.

The turbine inlet temperature should not be designed to be too high or too low; meanwhile the turbine inlet pressure should ensure that the working fluid before entering the turbine is saturated or superheated vapor. The condensation temperature should be as low as possible in a certain range that the heat transfer area is in proper size.

## 5. Conclusions

In this paper, the multi-objective optimization of a solar-powered CCHP system based on organic working fluid is conducted. The system operates in three operation modes, namely power mode, CHP mode and CCP mode. Turbine inlet pressure, turbine inlet temperature, condensation temperature and pinch temperature difference are selected as the decision variables, while the average useful output and the total heat transfer area are selected as the objective functions for the optimization. NSGA-II is employed to achieve the optimum performance in the multi-objective optimization.

The solutions of the optimization are obtained. For the power mode, the optimum average useful output and total heat transfer area are 6.40 kW and 46.16 m<sup>2</sup>. For the CCP mode, the optimum average useful output and total heat transfer area are 5.84 kW and 58.74 m<sup>2</sup>. For the CHP mode, the optimum average useful



output and total heat transfer area are 8.89 kW and 38.78 m<sup>2</sup>. Eventually, in the situation of this case, the turbine inlet temperature should be set to be a proper value based on the requirement that the turbine inlet pressure could ensure that the working fluid in the turbine inlet is saturated or superheated vapor. On the condition that the heat transfer area of the condenser is not too large, the condensation temperature could be as low as possible. The pinch temperature difference can be decided in different range for different modes. The optimal combination of the parameters in single objective optimization is different from that in multi-objective optimization under the given condition. Every point of the solutions in the multi-objective optimization is a candidate of the optimum result, implying that the multi-objective optimization provides a more comprehensive solution set so that the best performance can be achieved according to different requirements for system.

### Acknowledgments

The authors gratefully acknowledge the financial support by the National Natural Science Foundation of China (Grant No. 51106117).

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